



Heat Pipe Thermionic Reactor Concept

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Engineering Dept.

Risø-M-514

MASTER

HEAT PIPE THERMIONIC REACTOR CONCEPT

by

Erik Storm Pedersen

May, 1967

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Title and author(s)

Heat Pipe Thermionic
Reactor Concept

by Erik Storm Pedersen

Date

May 1967

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Abstract

The report describes a preliminary design study of a Heat Pipe Thermionic Reactor Concept and indicates a possible arrangement of a compact 1 MW(e) thermionic reactor.

In the Heat Pipe Thermionic Reactor Concept thermal power generated in the reactor core is transported by heat pipes to thermionic diodes located outside the reactor core. The thermionic emitters are in direct contact with the outside envelope of the heat pipes and the collectors are in contact with a liquid metal cooling system that transfers the waste heat to a radiator. The heat pipe uses the heat of vaporization and capillary action of a fluid to transport thermal energy at a high efficiency and a constant temperature along the entire heat pipe. It is therefore possible to obtain in-pile temperatures outside the core and thereby avoiding most of the problems associated with the in-pile thermionic designs, such as: Change of emitter dimensions due to fuel swelling upon irradiation; interaction between emitter and fuel; fission product contamination of the interelectrode space charge material; and the effect of nonuniform reactor power generation on the performance of the thermionic diodes.

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TABLE OF CONTENTS

	Page
1.0. INTRODUCTION	1
2.0. DESCRIPTION OF HEAT PIPE THERMIONIC REACTOR CONCEPT	2
3.0. HEAT PIPE PRINCIPLES	7
3.1. The Driving Pressure	9
3.2. Working Fluids	11
3.3. Properties of Lithium	12
4.0. PRELIMINARY DESIGN OF A 1 MW(e) HEAT PIPE THERMIONIC REACTOR	12
4.1. Thermionic Diode Design	12
4.2. Heat Pipe Design	14
4.3. Driving Force in Heat Pipe	16
4.4. Film Condensation	16
4.5. Boiling Heat Transfer	17
4.6. Wall Thickness of Heat Pipe	18
4.7. Weight Analysis	18
5.0. REFERENCES	20
6.0. ACKNOWLEDGMENT	21

LIST OF FIGURES

1. Heat Pipe Thermionic Reactor
2. Fuel Element With Heat Pipe
3. Thermionic Converter With Heat Pipe
4. Schematic Of Heat Pipe
5. Design Summary

1.0. INTRODUCTION

Direct conversion of heat into electricity has changed from a laboratory vision to that of working hardware. One of the main methods of direct energy conversion is thermionic energy conversion. Thermionic converters are compact, efficient, adaptable to a variety of heat sources, and capable, by multiple stacking, of producing a desired power level for extended periods of time. They operate at emitter temperatures from 1175°C to 2200°C , with individual power levels from a few watts to over 500 watts, efficiencies up to 20 per cent, and operation for periods in excess of 8000 hours out-of-pile.

One of the ideal heat sources for thermionic converters is the nuclear reactor. The thermionic converters may be placed inside (in-pile) or outside the reactor (out-of-pile). If they are inside, the fuel may either be used as the emitter itself or the emitter may be indirectly heated by the fuel. Most in-pile concepts have the fuel located inside the thermionic converter, however, one concept has the fuel surrounding the converter. If the thermionic converters are located outside the core, they may be arranged at the periphery of the reactor or they may be heated from liquid metal in an external loop where the liquid metal is heated by the reactor.

There are, however, many problems associated with these present designs. In the case of the in-pile design the major problems are the following: Change of emitter dimensions due to fuel swelling upon irradiation; interaction between emitter and fuel; fission product contamination of interelectrode space charge material; and the effect of nonuniform reactor power generation on the performance of the thermionic diodes. Another problem is the long times required for adequate testing of in-pile thermionic converters and converter components resulting in high costs. In the out-of-pile designs one is trying to get away from the in-pile problems by paying the price of lower performance and higher specific weights (lb/kW) of the overall power plant.

This Heat Pipe Thermionic Concept combines the most desirable features of both the in-pile and out-of-pile thermionic concepts and avoids most of the problems associated with these designs. Also, the concept clearly lends itself to external electrical heating of the heat pipe. By such a heating method, the complete heat pipe with the thermionic converter can be subjected to prolonged operational tests and repeated temperature cycling. Only those devices which pass with rigorous quality control tests would then be incorporated into the reactor.

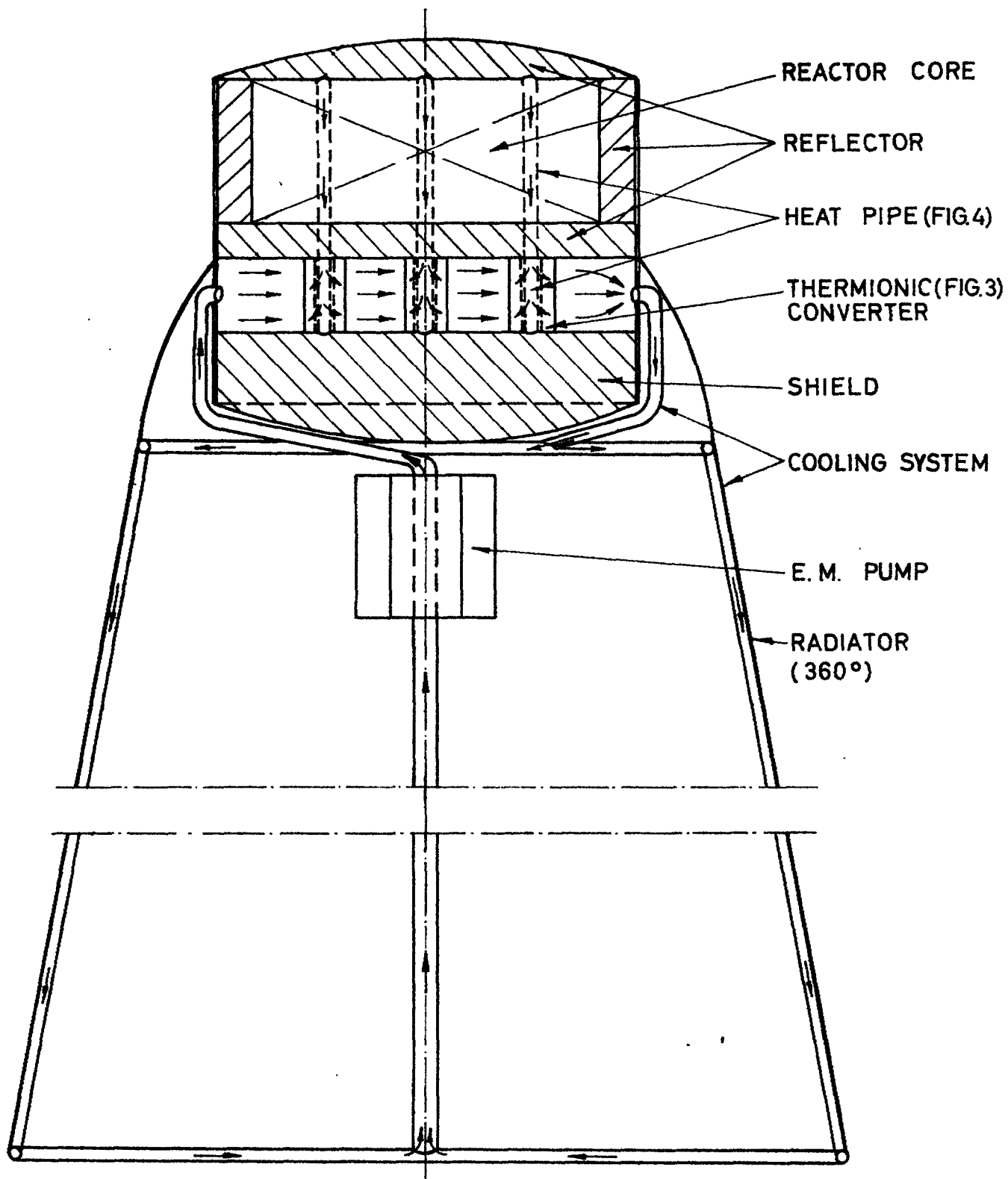
The Heat Pipe Thermionic Reactor Concept can be built with present day technology. The reactor may be a conventional thermal or fast reactor using suitable fuel, the heat pipe concept has been successfully tested by several laboratories and thermionic converters have obtained out-of-pile lifetimes in excess of 8000 hours. A preliminary system weight analysis indicate a specific system weight of about 12 lb/kW(e).

2.0. DESCRIPTION OF HEAT PIPE THERMIONIC REACTOR CONCEPT

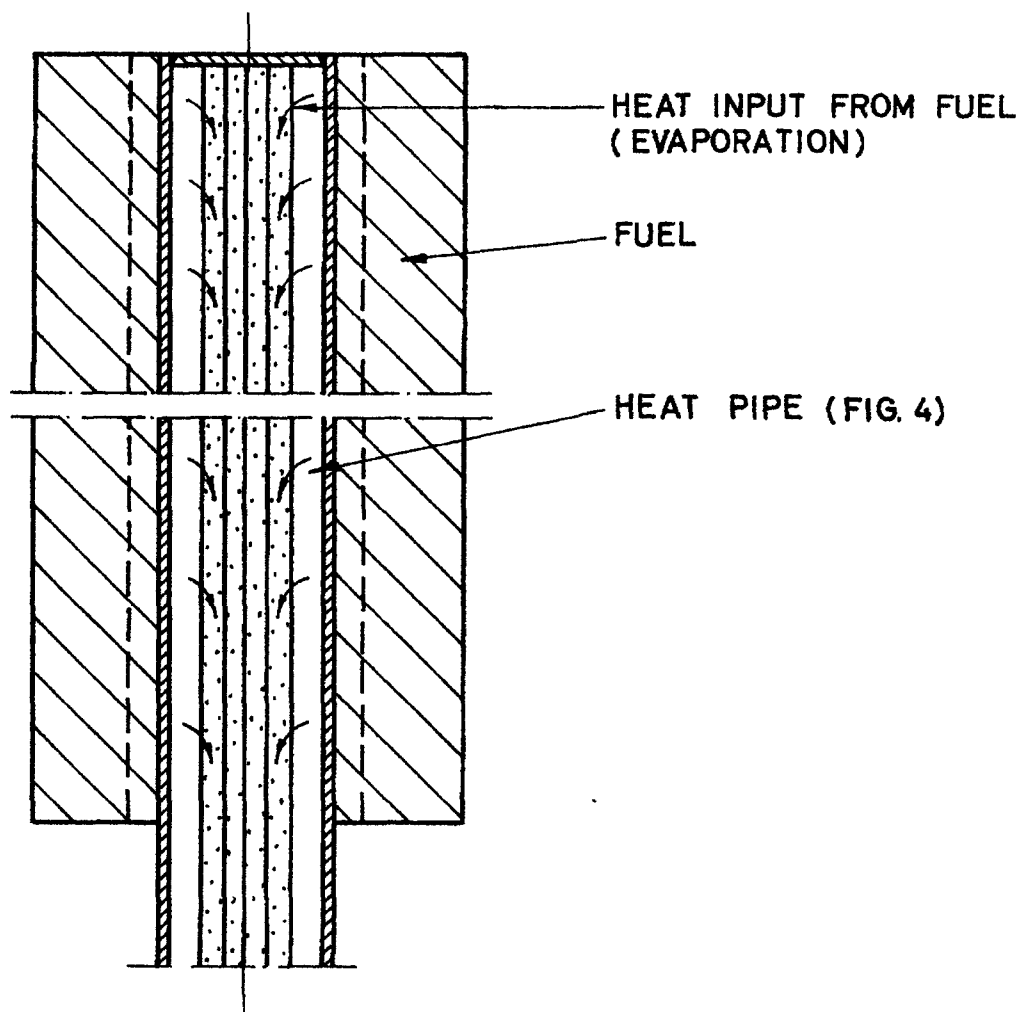
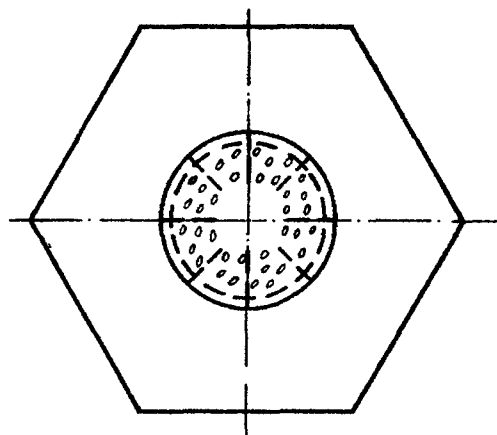
An outline of the Heat Pipe Thermionic Concept is shown in Figure 1. The main components are the following: Reactor core, heat pipe, thermionic converter, secondary cooling system, and a waste heat radiator. Thermal power generated in the reactor core is transported by heat pipes to thermionic converters located outside the reactor core behind a radiation shield. The thermionic emitters are in direct contact with the outside envelope of the heat pipes and the collectors are in contact with a liquid metal secondary cooling system that transfers the waste heat to a radiator. The cooling system utilizes an EM pump, however, it could also consist of a series of heat pipes transporting the waste heat from the thermionic collectors directly to the radiator. The reactor used for this concept may be a conventional thermal or fast reactor using suitable fuel. A possible arrangement of a fuel element with a built-in heat pipe is shown in Figure 2. As seen, the heat pipe is located in the center of the fuel block so that heat can flow into it from all directions. As the heat pipe operates at a low internal pressure its wall can be very thin. Also, during operation it will contain a working medium mainly in the form of vapor with a small neutron cross section. The heat pipe will, therefore, have little influence on the neutron flux and can be considered as a void in the core.

A possible arrangement of a thermionic converter built around one end of the heat pipe is shown in Figure 3. As seen, the diodes can be stacked on top of each other and connected in series. The diodes are insulated from the heat pipe and the secondary cooling system by a thin layer of electrical insulation. The power leads from each thermionic converter are connected in series and parallel to match the electrical requirements.

Preliminary design of a 1 MV(e) Heat Pipe Thermionic Reactor has been accomplished on the basis of using materials and fabrication

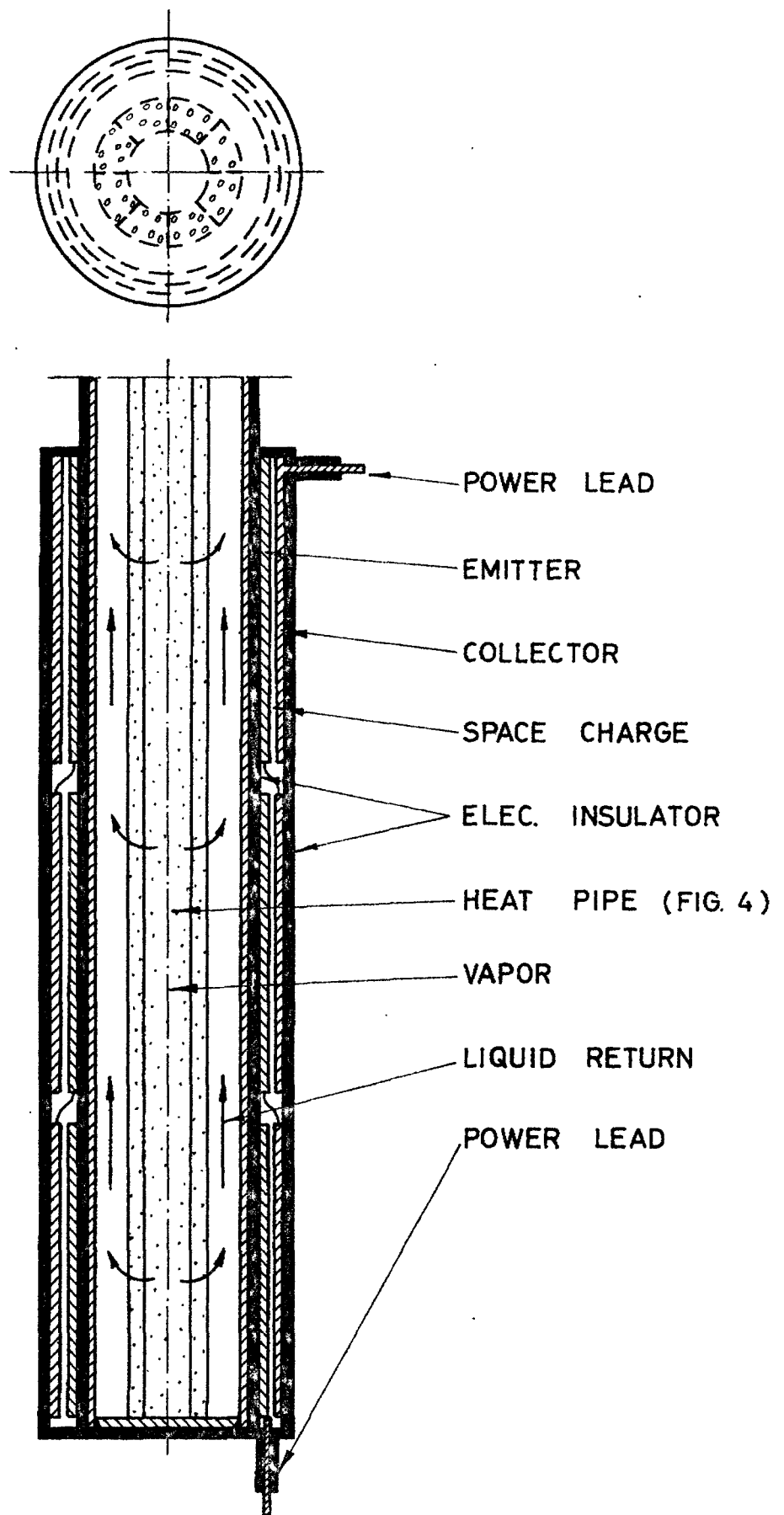


HEAT PIPE THERMIONIC REACTOR
FIG. 1



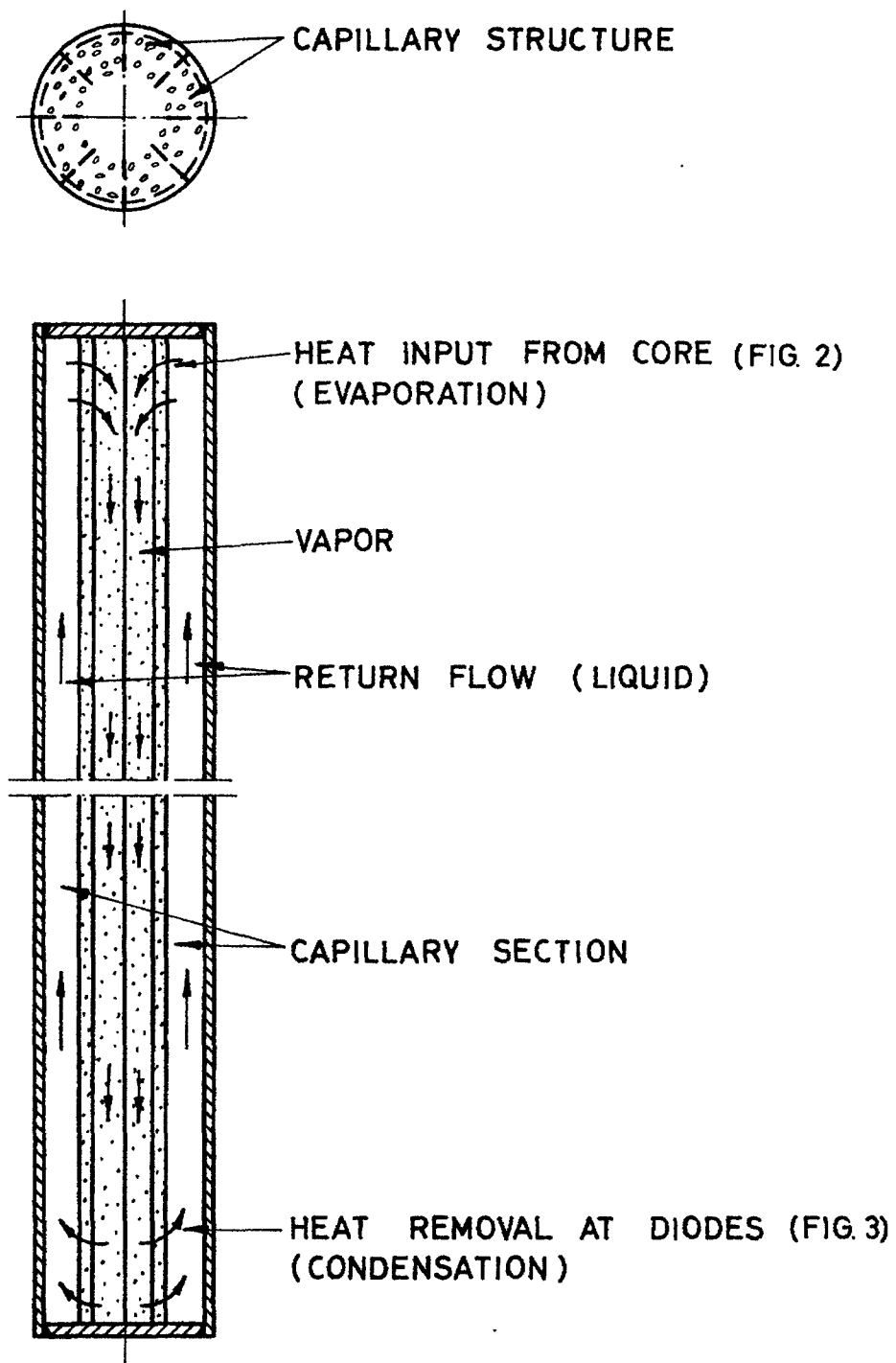
FUEL ELEMENT WITH HEAT PIPE

FIG. 2



THERMIONIC CONVERTER W/HEAT PIPE

FIG. 3



SCHEMATIC OF HEAT PIPE

FIG. 4

techniques available now or in the near future. The results of the preliminary design are given below.

Summary of Preliminary Performance Data:

Power Output	1000 kW(e)
Voltage Output	250 Volts
Current Output	4000 Amps.
Number of Thermionic Diodes	2500
Power Density (Thermionic)	$10 \text{ W}_e/\text{cm}^2$
Number of Heat Pipes	625
Size of Heat Pipe	2.5 cm Dia. x 54 cm lg.
Weight of Heat Pipes	210 kg
Core Diameter and Height	About 91 cm x 28 cm
Core Weight	1451 kg
U-235-Loading	573 kg
Fuel	$\text{UO}_2 + \text{W}$
Number of Fuel Elements	625
Mean Diameter of Fuel Element	3.64 cm
Fuel Volume Fraction	0.57

A preliminary system weight analysis gave the following results:

Summary of System Weights

Reactor	2100 kg
Shield	745 kg
Radiator	1250 kg
Heat Pipes	210 kg
Secondary Piping	150 kg
Secondary Pump	70 kg
Bus Bar	95 kg
Power Conditioning	<u>680 kg</u>
TOTAL =	5300 kg

or $\frac{5300 \text{ kg}}{1000 \text{ kW(e)}} = 5.3 \text{ kg/kW(e)} = 11.7 \text{ lb/kW(e)}$

This specific system weight of the Heat Pipe System compares very favorably with specific weights obtained in In-Pile thermionic systems.

A complete illustrative design summary of the Heat Pipe System is shown in Figure 5.

The thermionic reactor power system would have its highest voltage, if all 2500 diodes were arranged in series. However, failure of one diode would then shut down the entire system. A more desirable arrangement is to connect small groups in parallel, and then connect these groups in series with each other. In this manner, open-circuit failure of any one diode would be partially compensated by increased current densities through the diodes in parallel with it.

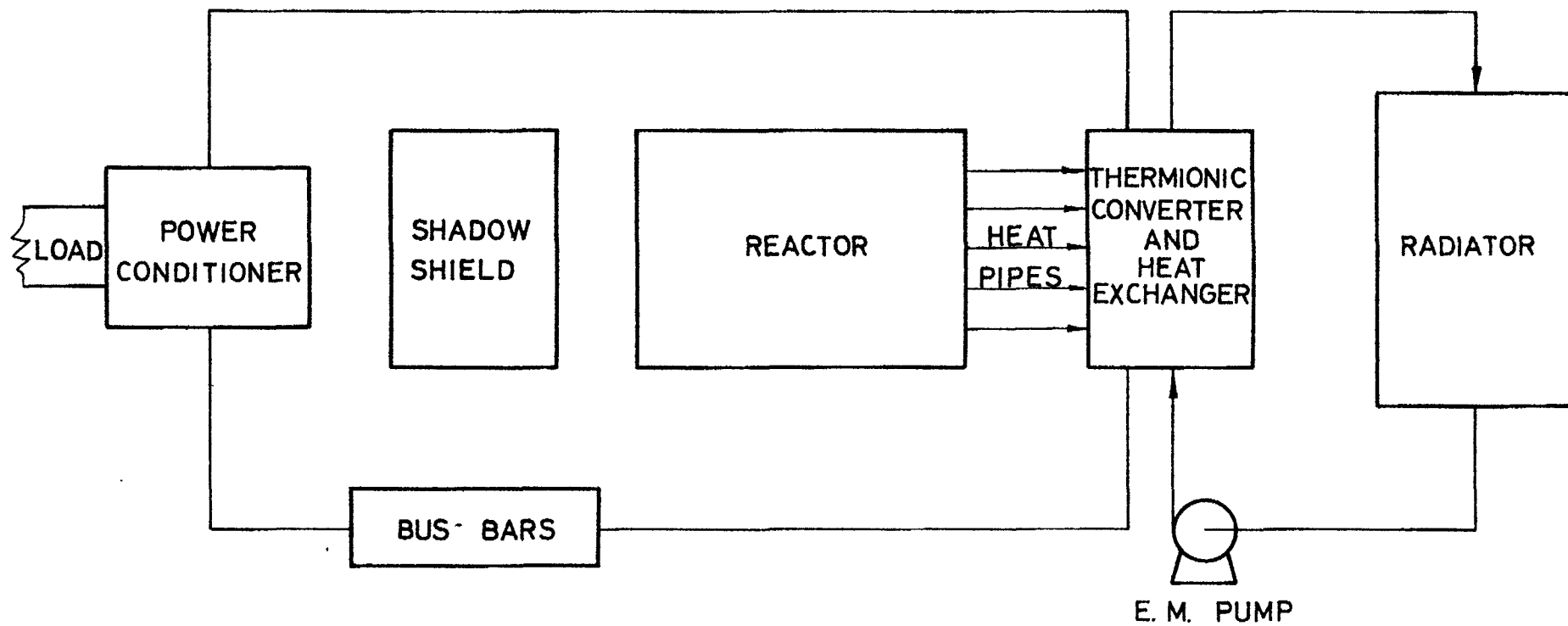
3.0. HEAT PIPE PRINCIPLES

The heat pipe is a static device consisting of a closed evacuated tube containing a small quantity of working fluid as shown in Figure 4. The tube incorporates a capillary structure along the inside walls and the means of accepting and rejecting heat at the respective ends. The heat pipe was originally developed by G. M. Grover and associates at the Los Alamos Scientific Laboratory. Other laboratories have also successfully tested the heat pipe concept.

The heat of vaporization and the capillary action control the operation of the heat pipe. About 4700 calories are required to boil one gram of Lithium to one gram of vapor with no change in temperature. Other liquids have similar characteristics, but the transfer takes place at different temperatures. This temperature can be changed by changing the pressure. The capillary action is the force that results from the relationship of the surface tension forces of the liquid and the material of the containment vessel which can drive the fluid through a wick or small diameter tubes.

The working of the heat pipe is shown in Figure 4. Heat is supplied at the input end imbedded in the reactor core and converts the liquid to a vapor by boiling. This action absorbs large quantities of heat energy and raises the temperature to the boiling point but no higher. The vapor is transported down the pipe to the heat removal end to equalize the energy levels. During this flow the heat pipe cannot change temperature along its length because any attempt at reduction will absorb large amounts of heat by condensation and thus maintain the level.

A constant temperature can therefore be maintained over the entire heat pipe within very narrow limits. The vapor is transformed to liquid by condensation at no change in temperature, when large amounts of



DESIGN SUMMARY

FIG. 5

heat is removed at the thermionic generator located at the output end of the heat pipe. The liquid now travels back to the evaporator or core at a rate determined by the capillary structure.

The heat pipe is a closed-circuit step system, and the step with the slowest rate of fluid transfer will determine the overall rate of flow and, therefore, heat transfer. Since evaporation produces a large change in volume for the elements of low molecular weight and since the viscosity of a gas is less than that of a liquid, different relative areas are needed for the transfer of liquid and vapor.

The heat pipe is intolerant of any temperature difference along its surface. It will remove energy by evaporation as required from any point where heat is available and deliver energy by condensation wherever a thermal drain exists. It will, therefore, accept heat readily from a non-uniform heat source and deliver its heat at a uniform temperature over the area of the heat delivery zone.

With the heat pipe it is possible to separate physically the heat input and heat removal zones such as the reactor core and thermionic generator. Thus, if the heat source and the heat-consuming device are themselves incompatible, the insertion of the heat pipe between them can make operation feasible.

3.1. The Driving Pressure

The driving pressure (by some authors called the driving force) is the difference between the vapor pressure in the evaporator P_2 and that in the condenser P_1 :

$$P_d = P_2 - P_1$$

To obtain an indication of how this driving pressure will arise or will be maintained in a dynamic system, we consider a cylindrical capillary tube with radius r_c .

In the static case there is a balance between the surface tension forces and the pressure in the liquid, so that a rise of the liquid can be observed in the tube, see Robert L. Daugherty and J. B. Franzini, Fluid Mechanics, p. 15. Assuming the meniscus is spherical and equating the lifting force created by surface tension to the gravity force, we write:

$$2\pi r_c \gamma \cos \Theta = \pi r_c^2 h \gamma$$

$$h = \frac{2 \gamma \cos \Theta}{\gamma \cdot r}$$

where

γ = surface tension in units of force per unit length

γ = specific weight of liquid

r_c = radius of capillary tube

h = capillary rise

Surface tension depends on the choice of materials, but also on the cleanliness of the surfaces as may be expected. It decreases slightly with increasing temperature.

For given conditions the meniscus is very stable, and it is not likely that its form will be changed to any extent by pressure differences in the system.

Assuming a capillary tube radius r_e at the evaporator end of the capillary system, the pressure within the liquid at the evaporator is:

$$P_e = P_2 - \frac{2 \gamma \cos \Theta}{r_e} = P_2 - \frac{2\gamma}{r_2}$$

where

P_2 = vapor pressure in the evaporator and

r_2 = meniscus sphere radius

and in the same way assuming a greater or different tube radius r_{con} at the condenser, the pressure within the liquid at the condenser is:

$$P_c = P_1 - \frac{2 \gamma \cos \Theta}{r_{con}} = P_1 - \frac{2\gamma}{r_1}$$

where

P_1 = vapor pressure in the condenser and

r_1 = meniscus sphere radius

If liquid is to flow from condenser to evaporator at a height $h_2 - h_1$ above it, the algebraic sum of the above-mentioned forces must be greater than zero to overcome the viscous drag in the fluid, so that

$$P_d = P_2 - P_1 = 2\gamma \left[\frac{1}{r_2} - \frac{1}{r_1} \right] - (P_c - P_e) - g \cdot \rho \cdot (h_2 - h_1) > 0$$

where

g = the gravitational constant

ρ = the density of liquid

If r_1 is great, this term may be neglected.

By designing a structure where P_d is greater than the viscous drag in the liquid, a flow from condenser to evaporator will be maintained. Experience has shown that this can be realised by choosing a capillary system of suitable dimensions.

A description of the first experiments performed by G. M. Grover, T. P. Cotter and C. F. Erickson, Los Alamos Scientific Laboratory, is published in Journal of Applied Physics 35, 2, 1964, p. 1990. These showed that a heat transfer of about 30 w/cm^2 could be obtained in a sodium system operating at a temperature about 1100°K .

3.2. Working Fluids

The various working fluids must be evaluated to select the one most suitable for the desired conditions of operation. The figure of merit of a fluid as a heat pipe medium varies directly with the boiling point temperature and inversely with its atomic weight. This is used in conjunction with standard vapor pressure curves for the various elements to determine the most suitable working fluid.

Lithium shows great promise for use with a high-temperature heat pipe system. It has a high boiling point (2430°F) and a low atomic weight (6.94). Also, its latent heat of vaporization (8338 BTU/lb) is the highest of the liquid metals. Lithium is a silver-white alkali metal. It is the hardest, least volatile, and least dense alkali metal. Among the alkali metals, it is the least reactive with oxygen and water. The corrosion properties of lithium are significantly different from those of the other alkali metals, being more similar to bismuth and lead in many respects. Columbium, tantalum, molybdenum and tungsten show relatively good resistance to lithium.

Some of the most important properties of lithium are listed below.

3.3. Properties of Lithium

<u>Property</u>	<u>Value</u>	<u>Temperature (°F)</u>
Physical:		
Atomic Weight	6.94	
Melting Point, °F	357	
Boiling Point, °F	2430	
Critical Point, psia	11850	4135
Density of Liquid, lb/ft ³	26.180	B. P.
Density of Vapor, lb/ft ³	0.00375	B. P.
Viscosity of Liquid, lb/ft-hr	0.40	M. P.
Surface Tension, lb/ft	0.027	M. P.
Thermal:		
Thermal Conductivity (Liquid), BTU/hr-ft-°F	26.54	M. P.
Thermal Conductivity (Vapor), BTU/hr-ft-°F	0.0459	B. P.
Specific Heat of Liquid, BTU/lb-°F	0.979	B. P.
Specific Heat of Vapor, BTU/lb-°F	0.708	B. P.
Latent Heat of Vaporization, BTU/lb	8430	B. P.
Latent Heat of Fusion, BTU/lb	186	M. P.

4.0. PRELIMINARY DESIGN OF 1 MW(e) HEAT PIPE THERMIONIC REACTOR

4.1. Thermionic Diode Design

The following assumptions are made:

$$T_e = 3292^{\circ}\text{F} \text{ (emitter temperature) } (2083^{\circ}\text{K})$$

$$T_c = 1452^{\circ}\text{F} \text{ (collector temperature) } (1050^{\circ}\text{K})$$

$$\phi_e = 3.8 \text{ volt (work function of emitter)}$$

$$\phi_c = 2.8 \text{ volt (work function of collector)}$$

The current emission is a function of the emitter as shown by the Richardson equation:

$$J_e = 120 \times T_e^2 \times \exp. \frac{-\phi_e}{K \times T_c}$$

where K = Boltzmann constant, 8.61×10^{-5} volts/ $^{\circ}\text{K}$

or

$$J_e = 120 \times 2083^2 \times \exp. \frac{-3.8}{8.61 \times 10^{-5} \times 2083} = 10 \text{ Amp./cm}^2$$

The output voltage is

$$V = \phi_e - \phi_c = 1.0 \text{ volt}$$

The output power density can now be calculated from:

$$P_{\text{out}} = J_e \times V = 10 \times 1 = 10 \text{ watts/cm}^2$$

The input power to the cell depends upon the losses such as: radiation loss, Peltier effect, conduction loss, and gas conductive loss. For the thermal losses we will use an effective emissivity of 0.13. The radiative power loss is therefore

$$P_r = E_r \times \sigma \times T^4 \text{ watts/cm}^2$$

where E_r = radiation emissivity, 0.13

$$\sigma = \text{Stefan-Boltzmann constant, } 5.71 \times 10^{-12} \text{ watts/cm}^2 - ^{\circ}\text{K}^4$$

or

$$P_r = 0.13 \times 5.71 \times 10^{-12} \times 2083^4 = 14.3 \text{ watts/cm}^2$$

The Peltier cooling power is

$$P_p = 10 \text{ amps} \times 3.8 \text{ volts} = 38 \text{ watts/cm}^2$$

Other losses are estimated at about 30% of the power output or 3 watts/cm². The expected efficiency of the thermionic converter is therefore:

$$\eta = \frac{\text{Power out}}{\text{Power in}} \times 100 = \frac{10 \times 100}{55.3} = 18.1\% \text{ or about } 18\%$$

The above results are in line with test results from RCA DIODE No. A1272. The thermionic diode has the following parameters:

Thermal Power Input	2170 W _{th}
Electrical Power Output	400 W _e
Load Resistance	0.0025 Ohm
Output Voltage	1.00 volt
Output Current	400 amps
Emitter Temperature	2063°K
Collector Temperature	1050°K
Cesium Temperature	608°K
Conversion Efficiency	18.3%
Emitter Surface Area	40 cm ²
Power Density (electrical)	10 W _e /cm ²
Power Density (thermal)	54 W _{th} /cm ²

4.2. Heat Pipe Design

Assume that total power output from Thermionic Reactor is 1000 kW_e. The number of diodes are therefore $\frac{1000 \text{ kW}_e}{0.4 \text{ kW}_e} = 2500$.

Assume that four diodes are required per heat pipe, we get the number of heat pipes equal to 625. No lead loss assumed. As there is one heat pipe per fuel element, we have 625 fuel elements in the core. The thermal power per fuel element is: $4 \times 2170 \text{ W} = 8680 \text{ W}_{th} = 8.2 \text{ BTU/sec}$. Length of Thermionic Converter is 20 cm

Core Data:

Core Diameter	91 cm
Core Height	27.5 cm
Core Volume	$1.3 \times 10^5 \text{ cm}^3$
Fuel Concentration, av. (62% UO ₂ + 33% W)	
U-235 Loading	573 kg

Assume Heat Pipe Diameter = 1.0 in. (2.5 cm)

Area of H. P. in Core = 216 cm²

Power Density in Core (H. P.) = 40 W_{th}/cm²

Volume of each Fuel Element = 283 cm³

Mean Diameter of Fuel Element = 3.64 cm

Fuel Volume Fraction = 0.57

Total length of Heat Pipe = 54 cm.

The minimum Lithium flow rate required in the heat pipe is

$$Q \text{ (BTU/sec)} = W \text{ (lb/sec)} \times h_v \text{ (BTU/lb)}$$

$$W = \frac{8.2 \text{ BTU/sec}}{8430 \text{ BTU/lb}} = 0.97 \times 10^{-3} \text{ lb/sec or } 0.06 \text{ cm}^3/\text{sec. (Liquid)}$$

Lithium vapor flow rate: $447 \text{ cm}^3/\text{sec}$

Vapor Area: 3.14 cm^2

$$\text{Vapor Velocity: } \frac{447 \text{ cm}^3/\text{sec}}{3.14 \text{ cm}^2} = 142 \text{ cm/sec.}$$

Liquid Area: 0.51 cm^2

$$\text{Liquid Velocity: } \frac{0.06 \text{ cm}^3/\text{sec}}{0.51 \text{ cm}^2} = 0.1 \text{ cm/sec.}$$

$$\text{Pressure Drop in Vapor Channel: } R_e = \frac{D \times V \times \rho}{\mu}$$

where $D = 2.0 \text{ cm} = 0.066 \text{ ft}$

$V = 1.42 \text{ m/sec} = 4.67 \text{ ft/sec}$

$\rho = 0.00375 \text{ lb/ft}^3$

$\mu = 0.0473 \text{ lb/ft-hr}$

$$R_e = \frac{0.066 \times 4.67 \times 0.00375 \times 3600}{0.0473} = 881 \text{ (Laminar)}$$

The pressure drop is

$$\Delta P = \frac{64}{R_e} \times \frac{\rho \times V^2 \times L}{2 \times g \times D} \text{ lb/ft}^2$$

where $L = 22 \text{ in.} = 1.8 \text{ ft.}$

$$\Delta P = \frac{64}{881} \times \frac{0.00375 \times 4.67^2 \times 1.8}{2 \times 32.2 \times 0.066} = 0.003 \text{ lb/ft}^2$$

or 0.01 mm Hg.

4.3. Driving Pressure in Heat Pipe

The driving pressure in the heat pipe is

$$F = 2 \times \gamma \left(\frac{1}{r_2} - \frac{1}{r_1} \right) - (P_c - P_e) - g \cdot \rho_L (h_2 - h_1)$$

where γ = surface tension

r_2 = radius of menisci at evaporator

r_1 = radius of menisci at condenser

P_e = pressure in liquid at evaporator

P_c = pressure in liquid at condenser

h = height

Assume

$$h_2 - h_1 \sim 0$$

$$P_c - P_e > 0 \quad (T_v = T_1 = \text{constant})$$

Diameter of each Capillary Tube ~ 0.5 mm

$$r_2 = 0.25 \text{ mm} \sim 0.0008 \text{ ft}$$

$$r_1 = 0.5 \text{ mm} \sim 0.0015 \text{ ft}$$

$$\gamma = 0.027, \text{ lb/ft}$$

$$F = 2 \times 0.027 \left(\frac{1}{0.0008} - \frac{1}{0.0015} \right) 0.87 = 31.5 \text{ lb/in}^3 = 11.1 \text{ mm Hg}$$

4.4. Film Condensation

Nusselt's result for heat transfer by condensation in a horizontal cylinder is:

$$h_m = 0.725 \times \left(\frac{L \cdot \rho^2 \cdot k^3 \cdot g}{\mu \cdot D \cdot \Delta T} \right)^{1/4}, \text{ BTU/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

where L = heat of vaporization, 8430 BTU/lb

ρ = condensate density, 26.18 lb/ft³

k = condensate thermal conductivity, 26.54 BTU/hr-ft-°F

μ = abs. viscosity of condensate, 0.4 lb/ft-hr

D = diameter of cylinder, 2 cm = 0.066 ft.

ΔT = wall-vapor tempt. difference $\sim 5^{\circ}\text{F}$

g = $32.2 \text{ ft/sec}^2 = 4.8 \times 10^6 \text{ ft/hr}^2$

$$h_m = 0.725 \left(\frac{8430 \times 26.18^2 \times 26.54^3 \times 4.8 \times 10^6}{0.4 \times 0.066 \times 5} \right)^{1/4} = 31600 \text{ BTU/hr-ft}^2\text{-}^{\circ}\text{F}$$

Check on ΔT :

$$\Delta T = \frac{Q/A}{h_m} = \frac{54 \text{ W/cm}^2 \times 3171.2 \frac{\text{BTU/hr-ft}^2}{\text{W/cm}^2}}{31600 \text{ BTU/hr-ft}^2\text{-}^{\circ}\text{F}} = 5^{\circ}\text{F}$$

4.5. Boiling Heat Transfer

From Liquid Metals Book we have

$$Q/A = 4.3 \times 10^{-5} \frac{(a_1 \cdot C_{pl} \cdot \rho_1 \cdot T_s)}{\gamma^{1/2} \cdot (\Delta L \cdot \rho_v)^{3/2}} (C_{pl} \cdot T_s \cdot a_1^{1/2})^{1/4} \\ \times (\rho_1/\mu_1)^{5/8} \cdot (P_{rl})^{1/3} \cdot \Delta p^2$$

where Q/A = heat flux, $1.71 \times 10^5 \text{ BTU/hr-ft}^2$

a_1 = thermal diffusivity = $k/C_p \cdot \rho = 1.04 \text{ ft}^2/\text{hr}$

ρ = density; $\rho_1 = 26.18 \text{ lb/ft}^3$; $\rho_v = 0.00375 \text{ lb/ft}^3$

T_w = surface temperature, 3600°R

T_s = saturation temperature, 3460°R

Δp = sat. pressure difference for $T_w - T_s$, lb/ft^2

ΔL = latent heat of vaporization, 8430 BTU/lb

μ_1 = viscosity, 0.4 lb/ft-hr

γ = surface tension, 0.027 lb/ft

C_{pl} = specific heat, $0.979 \text{ BTU/lb-}^{\circ}\text{F}$

k = thermal conductivity, $26.54 \text{ BTU/hr-ft-}^{\circ}\text{F}$

P_r = Prandtl No = $C_p \cdot \mu/k = 0.0147$

Solving for Δp , we get

$$\Delta p = \sqrt{\frac{1.71 \times 10^5}{0.532}} = 565 \text{ lb/ft}^2$$

$$\begin{aligned}\Delta p &= P_{tw} - P_{ts}; \quad P_{tw} = \Delta p + P_{ts} = 565 + 1.44 \times 10^4 \\ &= 1.5 \times 10^4 \text{ lb/ft}^2 = 104 \text{ psi}\end{aligned}$$

$$T_w \text{ (at 104 psi)} \sim 3020^\circ\text{F} \sim 3480^\circ\text{R}$$

$$\Delta T = T_w - T_s = 20^\circ\text{F}$$

$$h = \frac{Q/A}{\Delta T} = \frac{1.71 \times 10^5}{20} = 8.5 \times 10^3 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$$

4.6. Wall Thickness of Heat Pipe

The wall thickness is:

$$t = \frac{P \cdot R}{SE - 0.6P} \quad (\text{ASME CODE})$$

where P = design pressure, 100 psi

R = inside pipe radius, 0.45 in.

S = max. all stress, 2000 psi (tantalum) 10,000 (cb); 7000 (mo)

E = 1 (Seamless)

$$t = \frac{100 \times 0.45}{2000 - 0.6 \times 100} = 0.023 \text{ in.} \sim 0.058 \text{ cm}$$

Use 1 mm Wall Thickness (Tantalum)

4.7. Weight Analysis

Heat rejection from the radiator is

$$P = A \times \sigma \times \epsilon_r \times T_r^4 \text{ watts}$$

where A = radiator area

σ = Stefan-Boltzmann constant

$$= 5.71 \times 10^{-12} \text{ Watts/cm}^2\text{-}^\circ\text{K}^4$$

ϵ_r = radiator emissivity, 0.93

T_r = radiator temperature, 1000°K

$P = 5.48 \text{ MW}_{th} = 5.48 \times 10^6 \text{ Watts}$

Radiator Area:

$$A = \frac{5.48 \times 10^6}{5.71 \times 10^{-12} \times 0.93 \times 1000^4} = 1.03 \times 10^6 \text{ cm}^2$$

$$= 1.03 \times 10^2 \text{ m}^2 \times 10.764 = 1110 \text{ ft}^2$$

From other studies, we have the specific weight of a radiator = 2.50 lb/ft². Therefore, the total radiator weight is: 2.5 x 1110 = 2770 lb = 1250 kg.

The shadow shield weight is:

$$286 \text{ lb/ft}^2 \times (\pi/4 + (\frac{80.7}{2.5 \times 12})^2) = 1640 \text{ lb}$$

$$= 745 \text{ kg}$$

Summary of weights:

Reactor	2100 kg
Shield	745 kg
Radiator	1250 kg
Heat Pipes	210 kg
Secondary Piping	150 kg
Secondary Pump	70 kg
Bus Bar	95 kg
Power Conditioning	<u>680 kg</u>
TOTAL	5300 kg

or $\frac{5300 \text{ kg}}{1000 \text{ kW(e)}} = 5.3 \text{ kg/kW(e)} = 11.7 \text{ lb/kW(e)}$

This specific system weight compares favorably with specific weights obtained in in-pile thermionic studies.

5.0. REFERENCES

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